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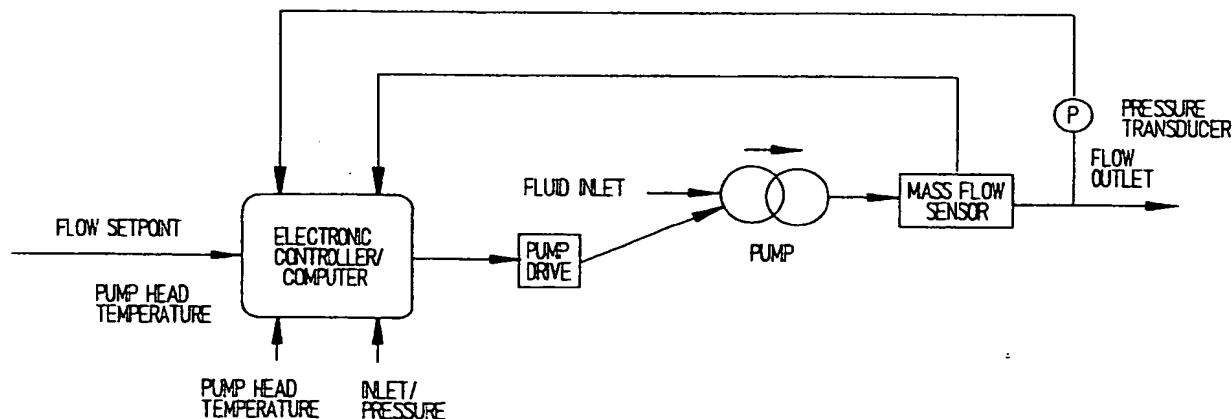
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W-8023 Pullach bei München(DE)(54) **Method and apparatus for real time compensation of fluid compressibility in high pressure reciprocating pumps.**

(57) The invention is a method and apparatus for significantly improving the performance of a reciprocating pump by delivering the pumping fluid at a desired pressure and flow rate with minimal flow fluctuations. The compressibility of the pumping fluid directly effects volumetric flow rate and mass flow rate. The method includes the step of sensing various pump parameters related to said compressibility and adjusting the pumping speed to make appropriate accommodations. In particular, the compressibility of the pumping fluid is effected by the adiabatic heating of the pumping fluid during compression, variations in pumping fluid density, leaks in the check valves of cylinder/piston seals and the primary and secondary switching losses in the fluid flow occurring when the primary and secondary pistons reverse direction.

**FIG 4A****BEST AVAILABLE COPY**

Background of the Invention

The invention relates to a pumping apparatus for delivering liquid at a high pressure, in particular a pumping apparatus for solvent delivery in liquid chromatography or supercritical fluid chromatography. The pumping apparatus pumps a mobile phase and a sample to be separated through the chromatographic system which includes a separation column and various transfer lines.

In chromatographic analysis, the flow rate of the liquid delivered to the column should be adjustable over a wide range. Once adjusted, it is very important to keep the flow rate constant as fluctuations cause variations in the retention time and associated areas under chromatographic peaks generated by chromatographic detectors of the separated sample. Flow sensitive detectors used in supercritical fluid or liquid chromatography include RI detectors, dielectric constant detectors, electrical conductivity detectors, NPD, ECD and fluorescence detectors. Since the area under the peaks are representative of the concentration of the chromatographically separated sample substances, fluctuations in the flow rate would impair the accuracy and the reproducibility of quantitative measurements. Additionally, many detectors are sensitive to flow and/or pressure ripples.

The compressibility of chromatographic solvents becomes noticeable at the high pressures encountered in high performance liquid chromatography and supercritical fluid chromatography. Such compressibility results in an additional source of fluctuations in the flow rate as the piston must compress the liquid to its final delivery pressure before actual delivery of liquid starts. This results in the outflow having pulsations which occur at the frequency of the pump. The percent magnitude of pulsations remains substantially constant over a wide range of flow rates, however, as the amplitudes of the peaks in the chromatogram become smaller at low flow rates, the influence of pulsations on the chromatographic results is more pronounced.

It is known to use a dual piston pump having two interconnected pump heads each with a reciprocating piston. The pistons may be driven via cams and a cam-shaft, a ball-screw drive or any other suitable mechanism which permits a phase difference resulting in a comparatively smooth outflow. A dual piston pump driven via cams and a common cam shaft is known from U.S. Pat. No. 4,352,636. One type of reciprocating pump which incorporates the ball-screw drive is disclosed in U.S. Patent No. 4,883,409 entitled "Pumping Apparatus for Delivering Liquid at High Pressure", and is hereby incorporated by reference. As set forth in Figure 1 the '409 patent discloses a pumping apparatus having two gear driven pistons which reciprocate in opposite directions in two pump chambers, respectively. The pistons are coupled to ball-screw drives which translate the rotary motion of the spindles into a linear motion of the pistons. The stroke volume can be changed by changing the amount in which the spindles are rotated during a pump cycle. Furthermore, a ball-screw drive permits the selection of any desired piston displacement over time during a pump cycle. For example, the displacement may be varied linearly as a function of time or accelerated for a short time to obtain a pre-compression phase.

Figure 2(a) illustrates the beginning of the intake cycle for a known ball-screw, dual chamber reciprocating pump illustrated in Figure 1 in which the output of the primary pump chamber is connected via a valve to the input of the secondary pump chamber. The stroke of each piston can be varied by controlling the angle of rotation of a reversible drive motor. The gear ratio is chosen to allow the primary piston to sweep twice as much volume as the secondary piston. If the primary and secondary cylinders have the same diameter, then the primary piston must move twice as fast as the secondary piston. Thus, if the primary piston has a volumetric stroke of L , the secondary piston will have a volumetric stroke of $L/2$. The outlet pressure can be kept constant at P_o using a back pressure regulator or a flow restrictor.

The pumping cycle for this pump is set forth in Figure 2(b). As the primary piston moves downward, the primary pressure decreases causing check valve 10 to open and check valve 20 to close, thus sucking in a fluid volume equal to L from the inlet. At the same time, the secondary piston delivers a volume of liquid ($L/2$) to the outlet. The pump motor then changes direction and the primary piston moves upward delivering a volume L . One half of this volume ($L/2$) is delivered to the outlet, and the other half is used to fill the vacuum generated in the secondary cylinder due to its downward motion. Assuming the fluid is incompressible, a flow of L/cycle is maintained in accordance to Figure 3(a)-3(d). The volumetric flow rate F_v for incompressible fluids can be expressed as follows:

$$F_v = L \times f_i \quad (\text{Volume per unit Eqn (1) time})$$

Where:

F_v = Volumetric flow rate at a defined outlet pressure

f_i = Pump frequency in cycles per unit time required to maintain flow rate of F_v when pumping

incompressible fluids.

$L =$ Primary piston stroke volume

The mass flow rate F_m can be expressed as:

$$F_m = \rho_i \times F_v = \rho_i \times L \times f_i$$

(Mass per unit Eqn (2) time)

Where:

$\rho_o =$ Fluid density at the outlet pressure. Assumed constant for incompressible fluids at all pressures and temperatures.

Since real fluids are compressible, Eqn 1 and Eqn 2 do not correctly set forth the volumetric flow rate and mass flow rate of a reciprocating pump. Furthermore, the compressibility of the fluid is a function of both pressure and temperature. Liquid chromatography and supercritical fluid chromatography systems typically treat these compressibility factors as constants, however, they vary considerably.

Summary of the Invention

In accordance with this invention, a method and apparatus are described for significantly improving the performance of a reciprocating pump by delivering the pumping fluid at a desired pressure and flow rate with minimal flow fluctuations. The method includes the step of sensing various pump parameters and adjusting the pumping speed to accommodate for the compressibility of the pumping fluid, the adiabatic heating of the pumping fluid during compression, variations in pumping fluid density, leaks in the check valves or cylinder/piston seals and the primary and secondary switching losses in the fluid flow occurring when the piston reverses direction. The compressibility of the pumping fluid directly effects volumetric flow rate and mass flow rate. These effects are much more noticeable when using supercritical fluids such as CO_2 than in other liquids. The assumption of a constant compressibility leads to optimal minimization of fluid fluctuation at only one point of the pressure/temperature characteristic. At other pressures and temperatures, flow fluctuations due to overcompensation or under-compensation becomes apparent. Accordingly, the object of the invention is to significantly reduce pumping and flow noise in reciprocating pumps by taking into account the pressure and temperature dependency of the compressibility of the pumping fluid.

The compression stroke of the pumping cycle is not isothermal as the rapid compression of the fluid causes a significant increase in fluid temperature. The density and the mass flow rate also vary with fluid temperature. This results in a positive flow pulse passing to the outlet and results in a flow ripple. To eliminate flow variations caused by temperature variations, the pumping fluid in the primary cylinder which is normally compressed to a volume C , is compressed to a different volume C' . The value of volume C' is larger than volume C and is calculated so that the fluid reaches the system pressure at the increased temperature and at a constant mass flow rate. Accordingly, it is another object of the invention to decrease the flow variations by taking into account the cyclic increase in temperature caused by the fluid compression.

The use of very low pumping flow rates is becoming more popular in liquid chromatography. When using a reciprocating pump at very small flow rates, leaks may form at the check valves or cylinder/piston seals. Identifying these leaks and compensating for them is extremely important when pumping supercritical fluid modifiers as it directly affects the mixing of the modifier in the CO_2 . Therefore, another object of the invention is to compensate for such leaks by measuring the flow with a high pressure flowmeter and increasing the pump speed of either the primary or the secondary piston or both pistons till the expected flow rate is achieved.

A secondary ripple in the fluid flow occurs when the piston reverses direction at the top of the stroke as some fluid escapes through the secondary check valve to the primary piston (secondary valve switching loss). It is another object of the invention to compensate for this secondary ripple by injecting a small volume of liquid in addition to the steady flow amount. The calculated value C' (new volume) is found to be larger from the measured empirical value due to the primary valve switching loss. The invention corrects the problem by continuously monitoring the pressure and adjusting the C' value to maintain a minimum pressure ripple. The difference between the calculated C' and the optimum measured C' establishes the switching loss in the primary valve. After the compression stroke, the fluid density in the primary cylinder increases as the fluid cools down to the temperature of the pump wall. This causes the mass flow and pressure to vary during the pumping cycle. Therefore, another objective of the invention is to modulate the pump speed so as to achieve constant mass flow rate and constant pressure.

Brief Description of Drawings

The present invention may be better understood, and its numerous objects and advantages will become apparent to those skilled in the art by reference to the accompanying drawings of which:

- 5 Figure 1 is a diagram of the Prior Art pumping apparatus. Figure 2(a) and Figure 2(b) are diagrams illustrating the primary and secondary piston positions during the pumping cycle.
 Figure 3(a), 3(b), 3(c) and 3(d) are fluid displacement and flow patterns through the pump for ideally incompressible fluids.
 Figure 4(a) and 4(b) are block diagrams of the invention in which different levels of sensing devices are
 10 employed for generating feedback signals.
 Figure 5(a), 5(b) and 5(c) are fluid displacement and flow patterns of a compressible fluid.
 Figure 6(a), 6(b) and 6(c) illustrates the flow pattern for compressible fluids in which compensation for compressibility is employed.
 Figure 7(a), 7(b) and 7(c) illustrate the flow pattern for compressible fluids in which compensation for
 15 compressibility is employed but with finite pump motor speed.
 Figure 8(a) and 8(b) represent the primary piston displacement and corresponding flow pattern observed with supercritical CO₂ with compression value calculated assuming isothermal compression. This is the actual output of a prior art pump which illustrates fluctuations in fluid flow.
 Figure 9(a) and 9(b) represent the primary piston displacement and corresponding flow pattern observed
 20 with supercritical CO₂ with adiabatic compensation C' and constant volumetric flow rate.
 Figure 10 is an actual pump output of Figure 9(b) showing flow fluctuations and ball screw noise.
 Figure 11(a) illustrates the relationship between primary piston displacement, constant volumetric flow rate and constant mass flow.
 Figure 11(b) illustrates the pressure and flow pattern for carbon dioxide using adiabatic compensation C' and programming volumetric flow so as to maintain a constant mass flow rate.
 25 Figure 12 is the actual pump output of Figure 11(b).
 Figure 13(a), 13(b), 13(c) and 13(d) illustrate very low flow patterns for the pump with an unbalanced at a constant outlet pressure and the pressure and flow pattern after leak correction.
 Figure 14(a), 14(b), 14(c) and 14(d) illustrate the pressure patterns for both a leaky primary and
 30 secondary when using a fixed restrictor and after leak correction.
 Figure 15 is the actual pump output of Figure 14(d).
 Figure 16 is a flow chart of the compression tracking algorithm.

Description of the Preferred Embodiment

35 In the preferred embodiment, a fluid pump as described in U.S. Patent No. 4,883,409 is modified to provide for sensing various pump parameters and adjusting of the pumping speed, such that fluctuations in the pumping pressure and mass flow rate are reduced.

In particular, Figure 4(a) illustrates the addition of mass flow sensors, pump head temperature sensors,
 40 inlet and outlet pressure sensors, all of which provide feedback to an electronic pump speed controller. The pump drive motor rotation angle, and the corresponding stroke of the two pistons (10,20), is controlled by firmware running on a microprocessor to accommodate for the compressibility of the pumping fluid, the adiabatic heating of the pumping fluid during compression, variations in pumping fluid density, leaks in the check valves or cylinder/piston seals and the primary and secondary switching losses in the fluid flow
 45 occurring when the piston reverses direction.

Figures 5 illustrates the flow patterns for compressible fluids as a function of primary piston displacement, flow through CV2 and the volumetric output flow F_v . In the primary delivery stroke, the primary piston moves upward to a calculable position C before the pressure in the primary cylinder increases from the initial intake pressure P_i to the outlet pressure P_o . A negative flow from the outlet, which is ideally
 50 maintained at a constant pressure, to the pump will occur during the compression displacement. The reduced average volumetric flow rate at an uncompensated outlet pressure F_v equals $C \times f$. In order to compensate for reduced and discontinuous flow, the pump motor is instructed to quickly compress the fluid upon intake by moving the primary piston twice the displacement needed to reach the outlet pressure using the maximum speed possible. As set forth in Figure 6, the primary piston will quickly move from position L
 55 to a new position defined as:

$$\text{Compensation position} = C - (L - C) = 2C - L \quad (\text{Eqn (3)})$$

Figure 6 (c) shows the resulting flow patterns. The pump frequency has been increased by a factor of L/C to make up for the reduced flow and is defined as:

New pump frequency $f = (L/C) \times f_i$ (Eqn (4))

By compensating for the fluid compressibility, the flow rate is returned to that of the incompressible fluid, where:

$$F_v \text{ (compensated)} = C \times f = C \times (L/C) \times f_i = L \times f_i \quad (\text{Eqn (5)})$$

In an ideal pump, the temperature of the pumping fluid can be assumed to stay constant when the primary piston moves from position L to position C . Additionally, the mass of the fluid in the primary cylinder stays constant such that:

$$(D + L) \times \rho_i = (D + C) \times \rho_o \quad \text{Eqn (6)}$$

Or:

$$\frac{D + L}{D + C} = \frac{\rho_o}{\rho_i}$$

Where:

D = Primary piston dead volume

L = Primary volumetric stroke

C = The primary piston position at which the primary cylinder pressure has just reached the outlet pressure P_o (unknown)

ρ_i = Fluid density right after the intake stroke (density at inlet pressure)

ρ_o = Fluid density right after the compression stroke (density at outlet pressure)

Both ρ_i and ρ_o can be obtained from the equations of state of the specific fluid employed. When compressed, the pumping fluid heats up and attempts to expand such that the outlet pressure is reached prior to the piston moving to the volume C . This effect causes a fast positive flow pulse to pass to the pump outlet (Figure 8b). Right after the compression stroke, the mass flow rate will be less than the calculated value due to decreased fluid density. The mass flow rate will continue to increase according to the time constant of heat exchange between the fluid and pump body. The resulting flow waveform due to this phenomena is shown in Figure 8B. This effect is more dramatic in fluids having high compressibility such as supercritical compressed gases. For supercritical carbon dioxide at outlet pressures above 200 bar, temperature rise of more than ten degrees centigrade have been calculated and empirically validated.

In order to overcome and virtually eliminate this large flow ripple, the fluid in the primary cylinder is compressed to a different volume C' . The value of C' is larger than C and is calculated so that the fluid reaches system pressure at an increased temperature. The primary piston is moved farther by the same displacement (to volume $2C' - L$) in order to fill the vacuum created in the secondary cylinder Figure 9A. For carbon dioxide with adiabatic compensation, this will result in the flow waveform shown in Figure 9(b) and the actual plot of Figure 10.

The pump speed is also programmed to ensure constant mass flow by accounting for the lost flow due to decreased density, and the elimination of flow ripple due to varying density. In particular, the post-compression temperature is first expressed as a function of time such that the fluid density can be represented as a function of time. From this equation, the piston speed required for constant mass flow can be derived.

Since the rapid compression of the pumping fluid causes the fluid to heat up and expand, the density decreases. When heat is transferred to the pump body, the pumped fluid cools and the density increases. The rate of density variation can be determined using the time function $f(t)$ of heat exchange between the fluid in the pump and its surroundings.

For constant mass flow rate:

$$\frac{d[V(t) - \rho(t)]}{dt} = -f$$

$$V(t) \frac{d\rho(t)}{dt} + \rho(t) \frac{dV(t)}{dt} = -f \quad \text{Eqn (7)}$$

P(t) can be approximated by the following function:

$$\rho(t) = \rho_i + (\rho_f - \rho_i) (1 - e^{-t/\tau})$$

ρ_i initial fluid density right after the fast compression stroke
 ρ_f final density at steady state pump temperature and outlet pressure
 t time
 $\rho(t)$ fluid density as function of time
 $V(t)$ pump volume filled with fluid at outlet pressure
 τ thermal time constant of heat exchange between the fluid and the pump walls. ~ 3 seconds
 f required mass flow rate

Solve Eqn (7) for

$$\frac{dV(t)}{dt}$$

which is the volumetric flow rate (piston speed)

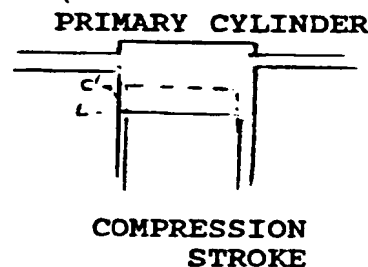
For any fluid, entropy "s" is a function of pressure and temperature. This can be expressed as:

$$s = s(P, T) \quad \text{Eqn (8)}$$

Therefore a constant entropy curve can be expressed as:

$$s(P, T) = \text{constant} \quad \text{Eqn (9)}$$

Fluid temperature and pressure at point L are known (pump head temperature and inlet pressure).
 At point C', pressure is equal to outlet pressure and temperature is unknown.



Thus, we can write:

$$s(P_o, T_c) = s(P_i, T_i) \quad \text{Eqn (10)}$$

The only unknown in Eqn B is the compression temperature T_c' . After solving for T_c' , one can substitute T_c' in the fluid equation of State: $P = P(\rho, T)$ Eqn (11)

and find the new density at this point C' (ρ_c'). Density and entropy look-up tables for the specific fluid can be used as well as the equations of state for this fluid. Using a "universal" equation of state may be preferable as it can apply to many different fluids and can simplify computer mathematical manipulations.

The value of C' can be found from the equation:

$$\frac{D + L}{D + C'} = \frac{P_c}{P_L} \quad \text{Eqn (12)}$$

Figure 11(a) illustrates the relationship between primary piston displacement, constant volumetric flow rate and constant mass flow rate. Figure 11(b) illustrates the flow pattern for carbon dioxide using C' and programming volumetric flow so as to maintain constant mass flow rate.

At very low flow rates, leaks may form through check valves or cylinder/piston seals. Absolute flow correction can be made by measuring the flow rate with a high pressure flowmeter and increasing the pump speed till the expected flow rate is achieved. If either side is leaking more than the other, then the flow will be uneven during each associated pump cycle. This leads to flow ripple as illustrated in Figure 13(b) and Figure 13(c). This ripple can be minimized by increasing the speed of the piston on the side which is leaking until a steady flow is obtained (Figure 13(d)). In addition to smoothing out the ripple, this technique also corrects the absolute flow as the actual flow of the more leaky side is increased to that of the less leaky side.

In another embodiment which does not use a high pressure flowmeter, the differential leak flow ripple can be minimized by pumping fluid into a fixed restrictor whose value is chosen to obtain about the same outlet pressure and the required flow rate. Figures 14(b) and 14(c) illustrate the pressure patterns for both a leaky primary and secondary when using a fixed restrictor. The outlet pressure is monitored and analyzed by a microprocessor to indicate which side is leaking the most such that the corresponding piston can be moved faster to correct for the pressure ripple. Figure 14(d) illustrates the pressure and flow patterns after leak correction. Figure 15 is the actual plot of Figure 14(d).

Secondary ripple is a negative ripple due to the flow lost when the pistons change direction as the secondary starts pumping, and is particularly apparent with small flow rates. Injecting an additional volume of liquid during the pumping cycle compensates for this secondary ripple and smooths out the flow rate. The injection is performed by setting the pump speed to the maximum speed allowed for a very short time right after the piston changes direction. The amount of fluid mass to be injected can be estimated using the area of the negative ripple in the flow diagram. A simplified block diagrams illustrating a feedback loop incorporating a pressure and flow transducer are set forth in Figure 4. While examining the output of the mass flow sensor or the pressure transducer, adjustments can be made to the pump drive to minimize the flow ripple.

The invention also provides for automatically compensating for primary valve switching loss by continuously monitoring the pressure waveform and changing compressibility compensation to maintain the desired pressure. By analyzing the pressure waveform, firmware can detect under-compensation and over-compensation and increase or decrease the amount of compensation accordingly. Figure 16 is a flowchart illustrating how the system works, and in particular, the following three general steps required for automatic compensation of primary valve switching loss:

- 1) Initially, the empirical compensation C_e is set to equal the theoretically calculated value C' .
- 2) A volumetric "window unit" is entered, for example, 0.1 micro-liter.
- 3) A control loop is entered and repeats itself every cycle starting after the compression stroke.

The control loop includes a system which diverges if neither of the tracking or converging flags are set and there is no compensation reversal (this indicates that the direction of the compensation has changed from the last cycle). In this case, the window is opened up to twice its previous size. Compensation C_e is then increased or decreased by the window size if the system was under-compensated or over-compensated respectively.

The system is determined to be converging on the target if there is compensation reversal and the window size is greater than one. In this case the window gets closed to half its previous size. Compensation C_e is increased or decreased by the window size if the system was under-compensated or over-compensated respectively. The converging flag is then set.

The system is determined to be on track of the target if there is compensation reversal and the window size is set to one. In case of over compensation, the compensation value C_e is decreased by one window unit. In case of under compensation, C_e is increased by one window unit. Tracking flag is set.

If the tracking flag is set and there was no compensation reversal, the flag is cleared and the window size is left unchanged in order to smoothly control slow target drift while tracking.

If the converging flag is set and there was no compensation reversal, the flag is cleared and the window size is left unchanged in order to converge successfully.

Although the best results are obtained by the forgoing pumping method and apparatus, changes and modification of the invention, as set forth in the specifically described embodiments, can be carried out without departing from the scope of the invention which is intended to be limited only by the scope of the appended claims.

Claims

1. A multiple cylinder reciprocating pumping apparatus for delivering a liquid or supercritical fluid at a desired pressure and flow rate with minimal fluctuations, comprising:

a first and a second pumping chamber, each having an inlet valve and an exit valve, wherein fluid to be pumped by the pumping apparatus is introduced into the inlet valve of said first pumping chamber and wherein fluid pumped by said pumping apparatus is expelled from the exit valve of said second pumping chamber; and wherein said first and second pumping chambers are coupled to each other through said first exit valve and said second inlet valve; and wherein, said first pumping chamber has a greater pumping capacity than said second pumping chamber; and

a first and second piston which reciprocate out of phase in said respective first and second pumping chambers; wherein said fluid is drawn into said first pumping chamber when said first piston is lowered, closing said check valve between said first and second chamber, while said second pumping chamber is expelling pumping fluid, and wherein said fluid both fills said second chamber and is expelled from said second chamber through said exit valve, when said first piston is raised and the second piston is lowered, as the first chamber has a greater pumping capacity than said second chamber; and

drive means for actuating said pistons; and

sensing means coupled to said pumping chamber to generate a temperature signal and a pressure signal corresponding to the temperature and pressure of the pumping fluid; and

a controller for generating drive signals responsive to said temperature and pressure signals and coupled to said drive means for controlling the speed and position of said first and second pistons to minimize pulsations in the fluid flow exiting the pumping apparatus, wherein, at the beginning of the pumping cycle, said drive means is instructed to rapidly compress the pumping fluid to the desired pressure and flow rate while compensating for the compressibility of the pumping fluid.

2. The pumping apparatus as claimed in claim 1, wherein said controller generates drive signals to compensate for the decrease in density of the pumping fluid caused by increased temperature upon compression.

3. The pumping apparatus as claimed in claim 1, wherein said sensing means generates a fluid leak signal in response to fluid leaks at the check valves or cylinder/piston seals, wherein, said controller generates drive signals in response to said fluid leak signal, for increasing the pumping speed to maintain the desired flow rate and pressure.

4. A multiple cylinder pumping apparatus for delivering a liquid or supercritical fluid at a desired pumping pressure and flow rate with minimal fluctuations, comprising:

a plurality of cylinders and associated pistons;

drive means coupled to said pistons for actuating said pistons in accordance to said control signals

sensing means for sensing pump parameters and generating feedback signals corresponding to said sensed parameters;

a controller coupled to said sensing means and said drive means for generating drive signals in response to said feedback signals to control the actuation of said pistons, wherein, said pumping apparatus delivers said liquid or supercritical fluid at a desired pumping pressure and flow rate by adjusting the actuation of said pistons to compensate for said sensed parameters.

5. The apparatus of claim 4, wherein said sensed pump parameter is the temperature of the fluid within said cylinder and is employed for calculating the density of said fluid such that the pump delivers the pumping fluid at a constant mass flow rate.

6. The apparatus of claim 5, wherein said temperature is periodically sensed such that said controller can periodically generate drive signals to compensate for variations in fluid density.

7. The apparatus of claim 4, wherein said sensed pump parameter is the pressure of the fluid at the output of the pump, and pressure measurements which are lower than expected values a leak in the pump, and wherein, said controller generates drive signals to increase the speed of primary or secondary pistons or both pistons to compensate for said leaks.

8. The apparatus of claim 7, wherein variations in fluid flow rate or pressure indicates the presence of secondary valve switching loss, and wherein, said controller generates drive signals in response thereto such that an additional amount of pumping fluid is injected to maintain a minimum pressure ripple.

9. The apparatus of claim 8, wherein the primary valve switching loss is compensated for by increasing the compression stroke and the said loss is automatically tracked.

10. A method for delivering a pumping fluid, such as a liquid or supercritical fluid, at a desired pressure and mass flow rate with minimal fluctuations, comprising the method steps of:

- sensing the temperature and pressure of the pumping fluid during and the pumping cycle; and
- adjusting the pumping speed to compensate for density changes in the pumping fluid resulting from said temperature and pressure changes such that the fluid is pumped at a desired pressure and mass flow rate with minimal fluctuations.

11. The method of claim 10, further comprising the step of compensating for leaks in the cylinder/piston seals by increasing the pumping speed of the leaking cylinder/piston, said leaks being identified by the method steps of

- sensing the output flow or pressure waveform; and
- calculating which cylinder/piston is leaking such that the corresponding pumping speed can be increased.

12. The method of claim 11, further comprising the step of compensating for and auto-tracking the valves switching losses.

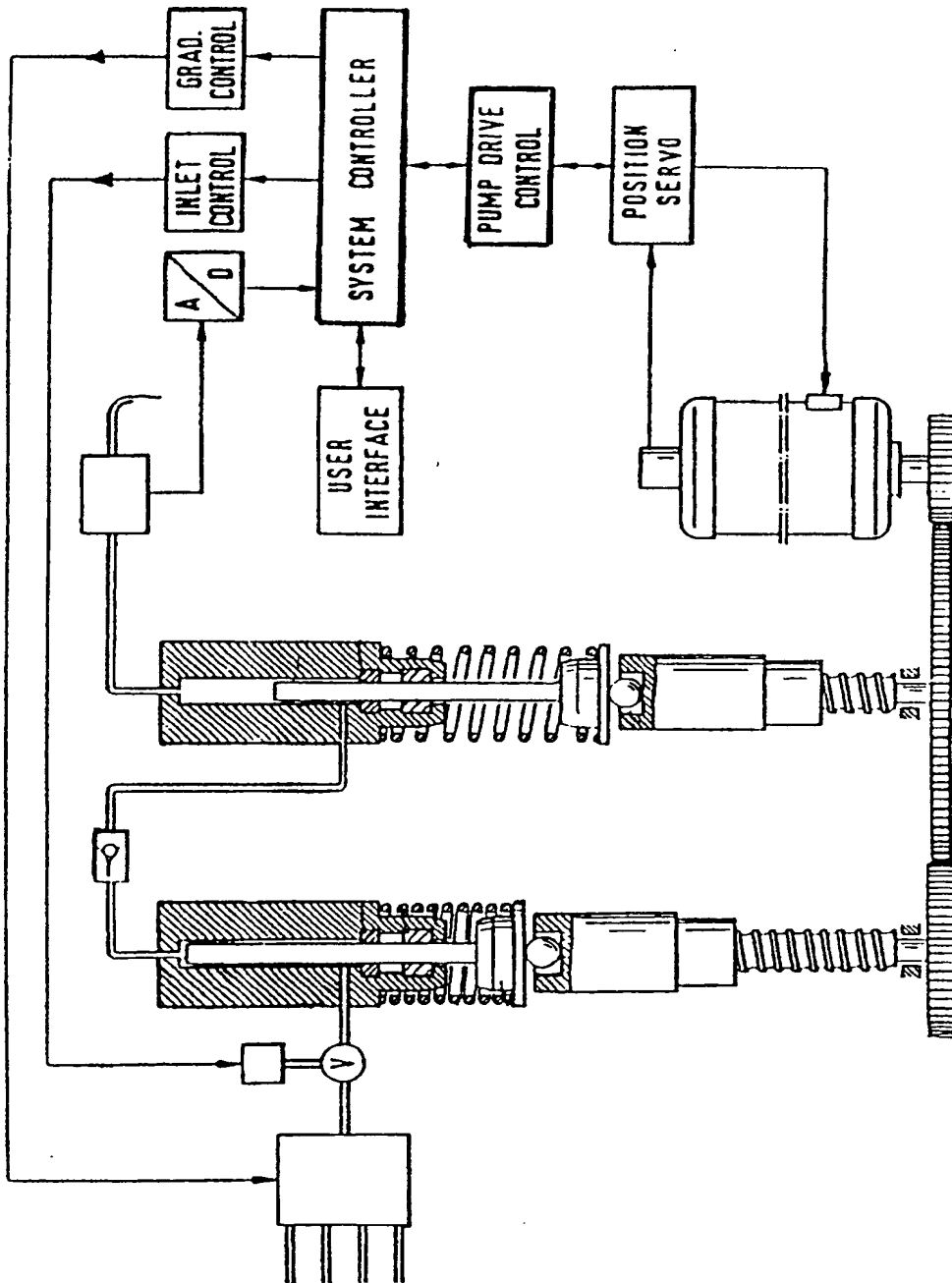


FIG 1 PRIOR ART

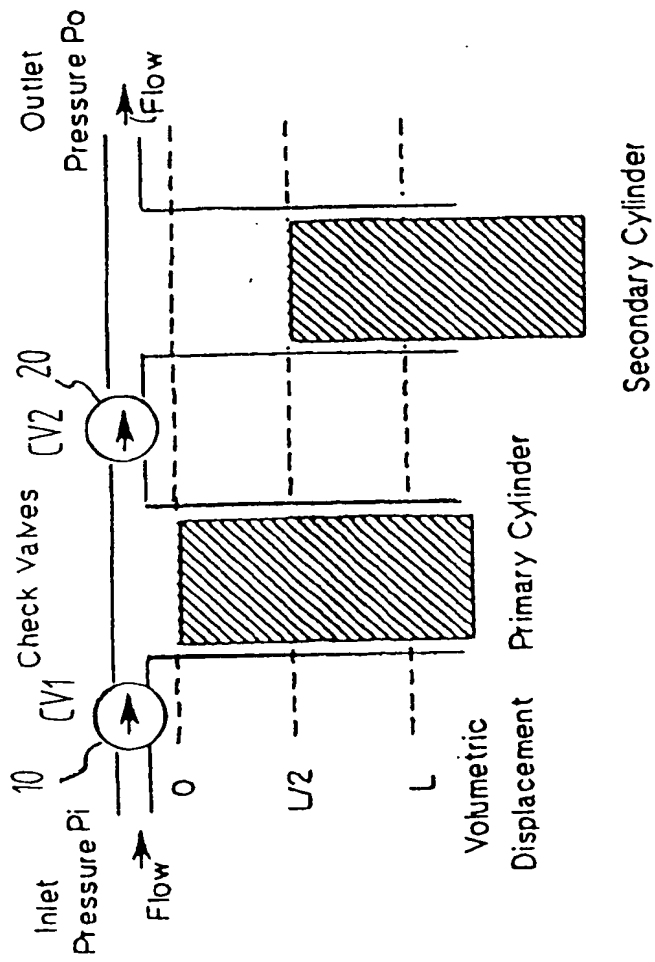


FIG 2A PRIOR ART

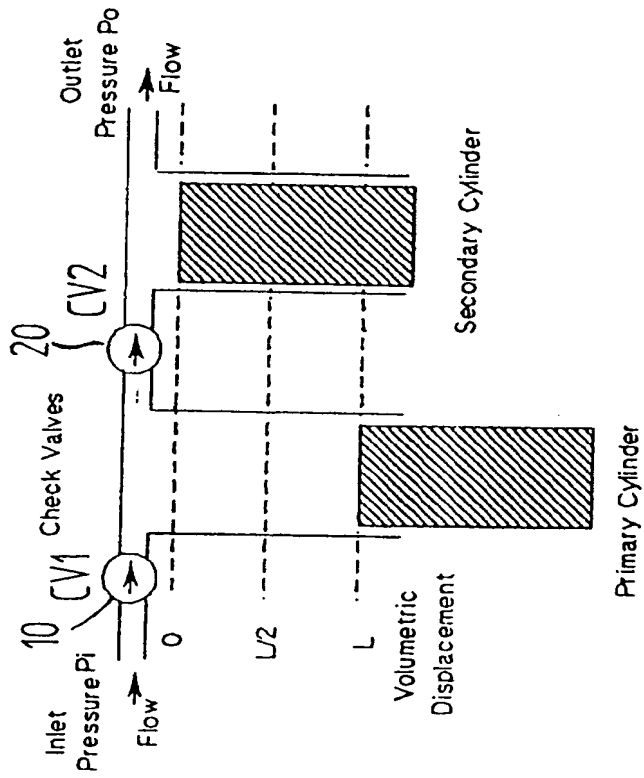
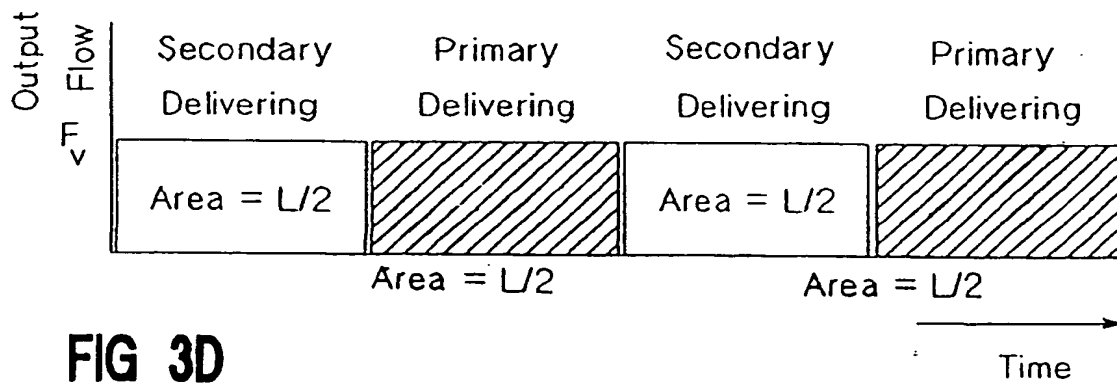
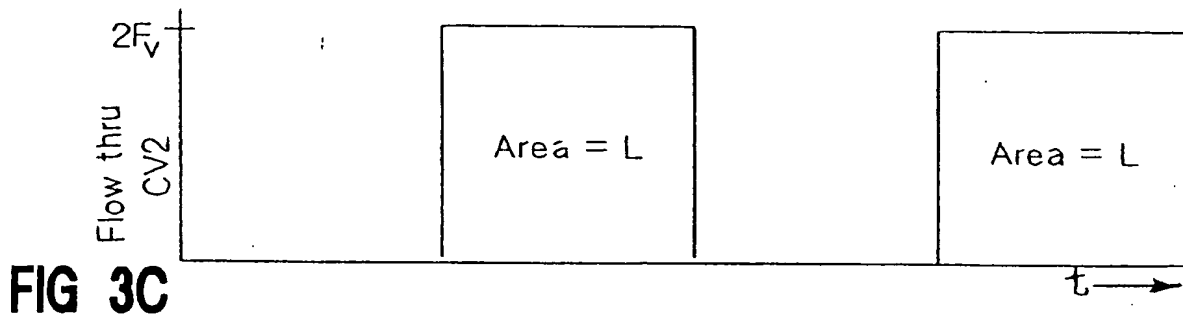
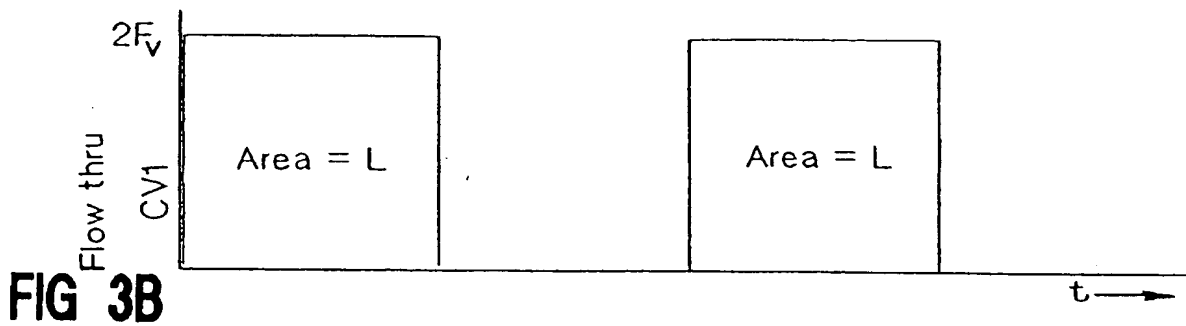
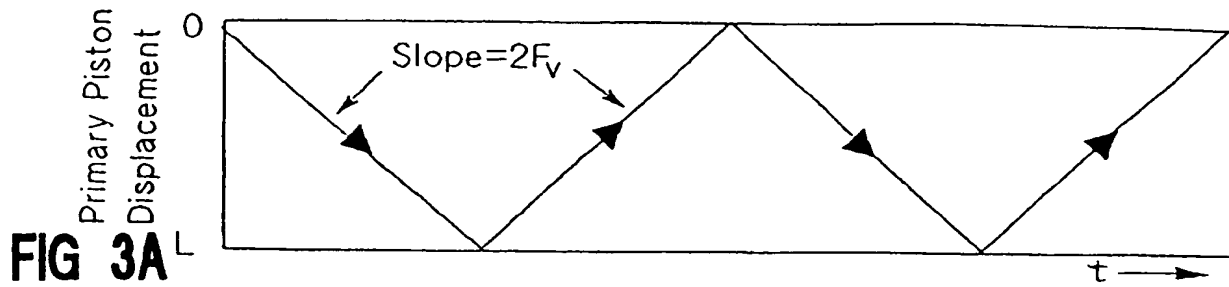


FIG 2B PRIOR ART



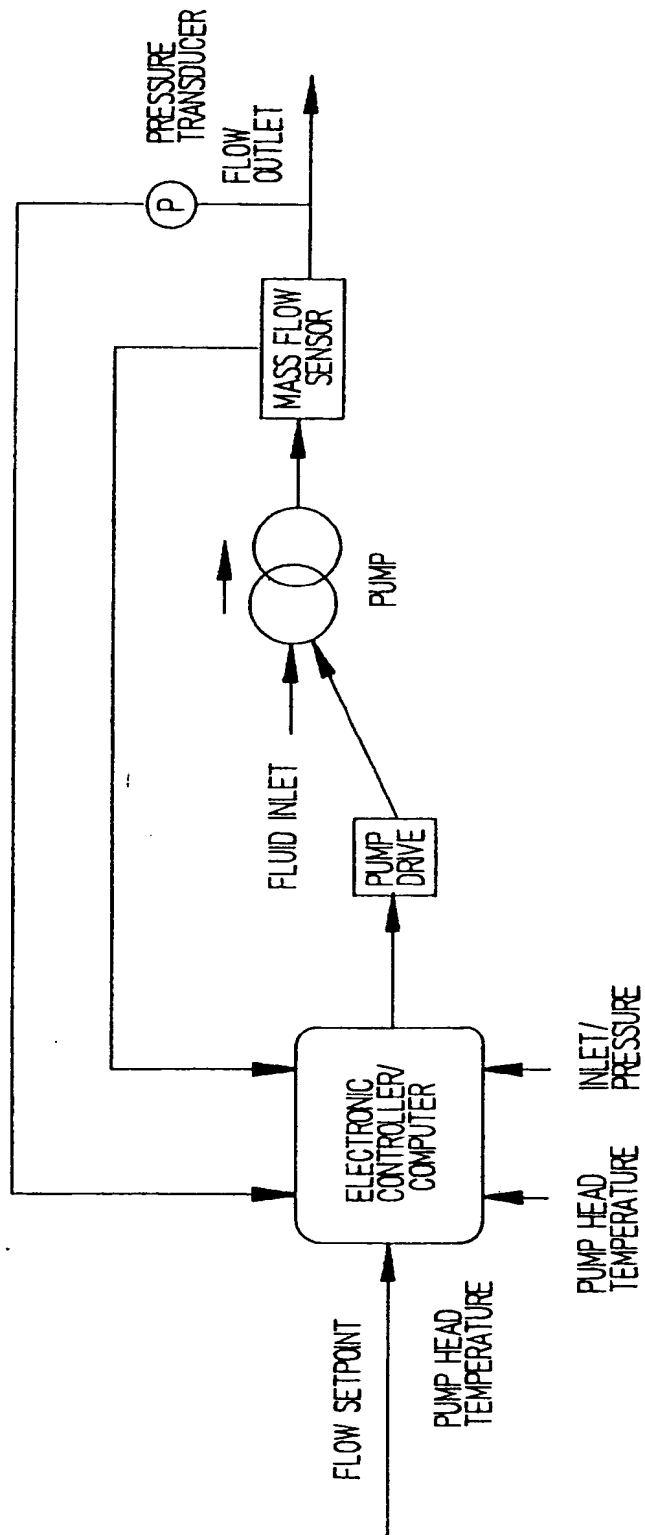


FIG 4A

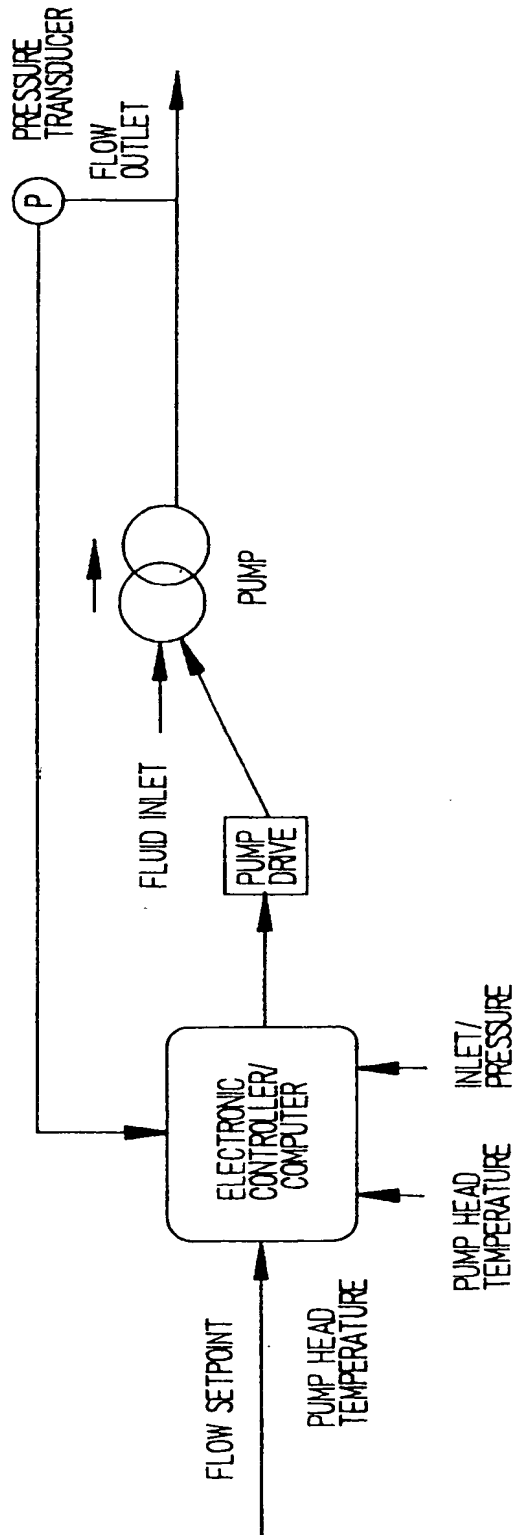
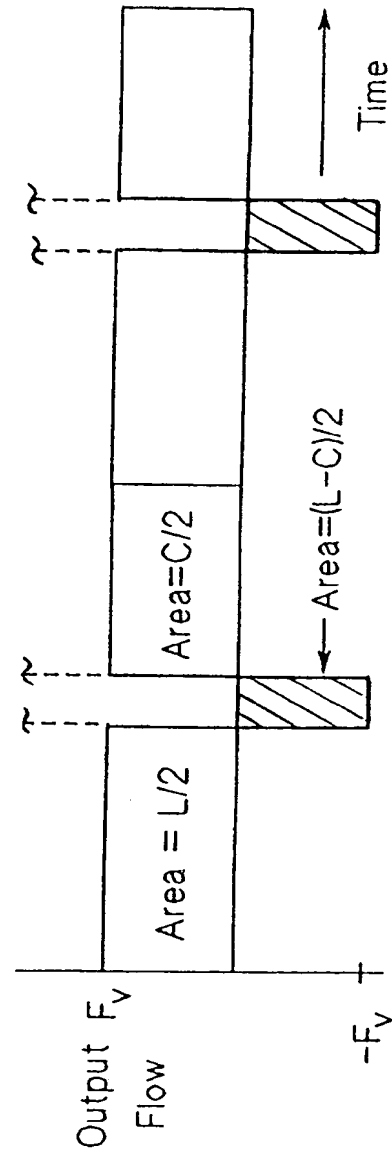
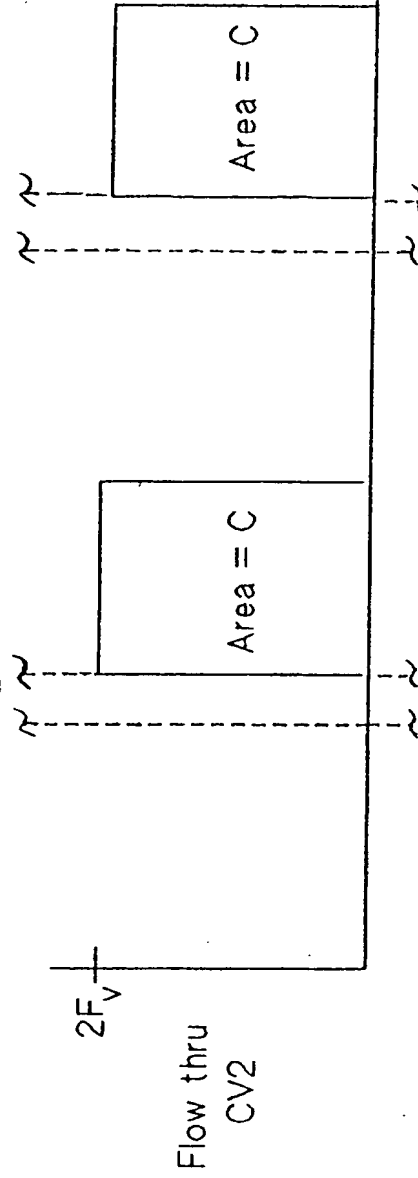
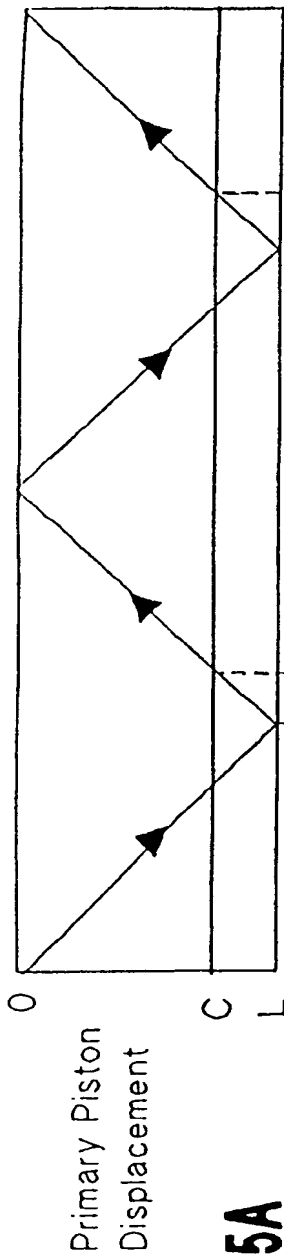


FIG 4B



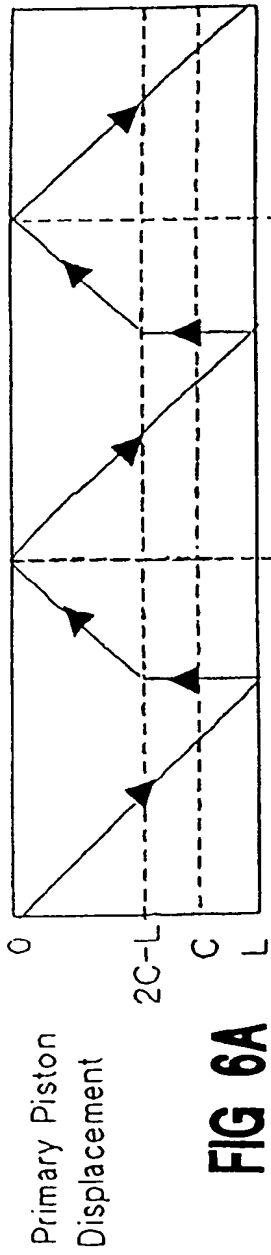


FIG 6A

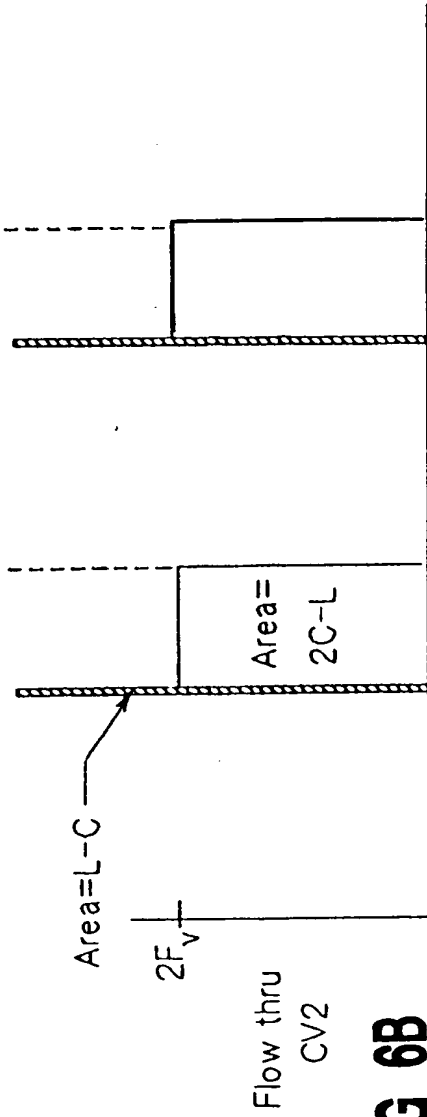


FIG 6B

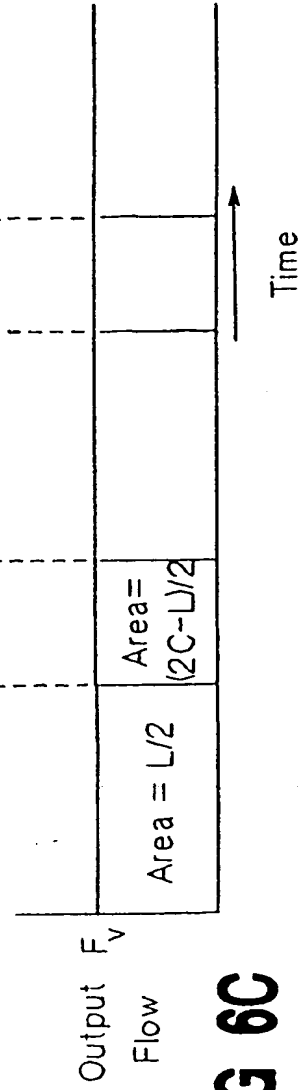


FIG 6C

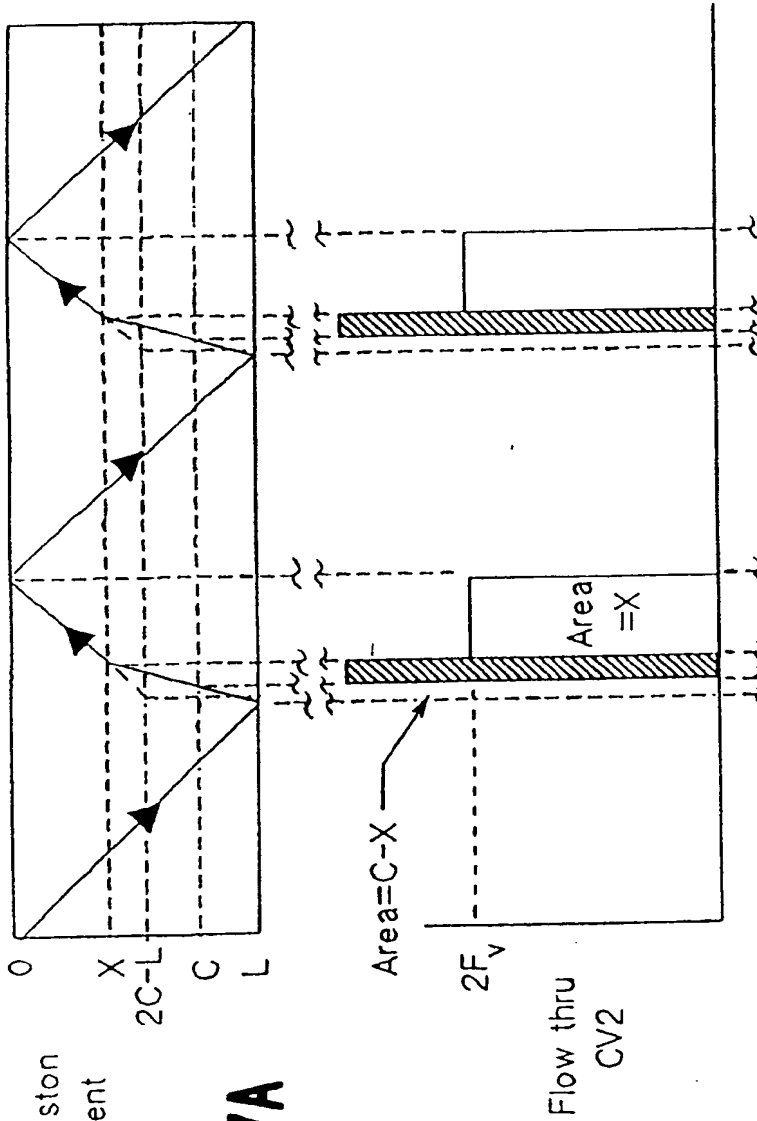
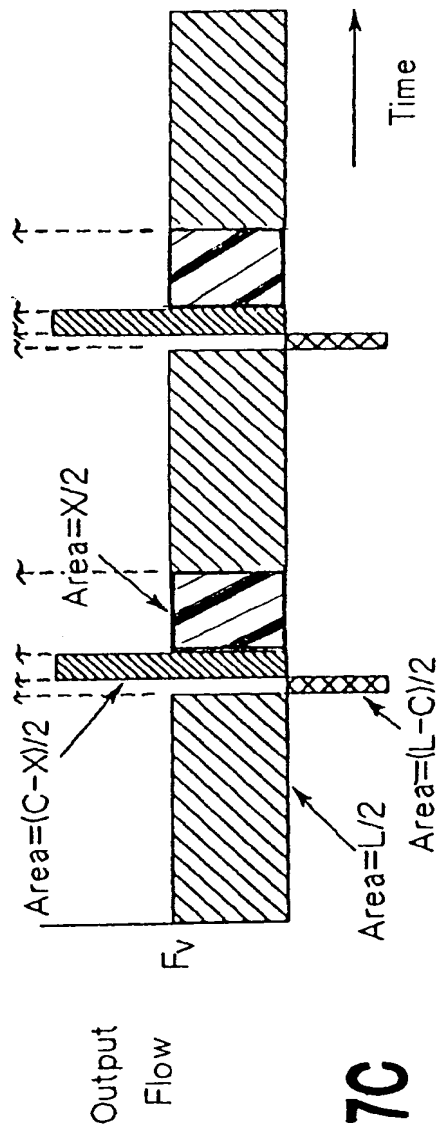


FIG 7B



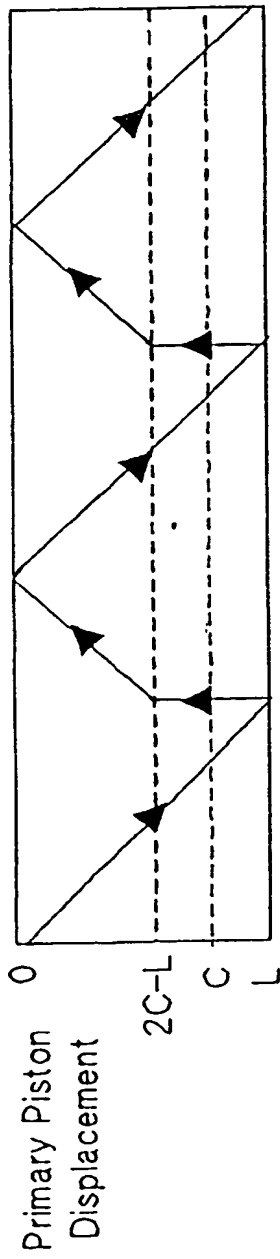


FIG 8A

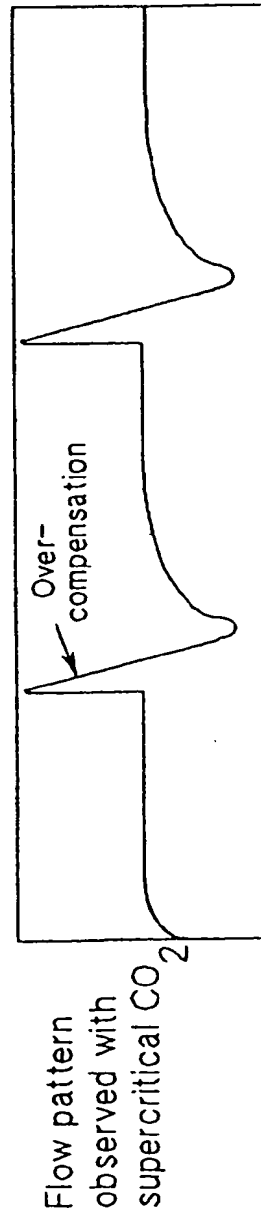


FIG 8B

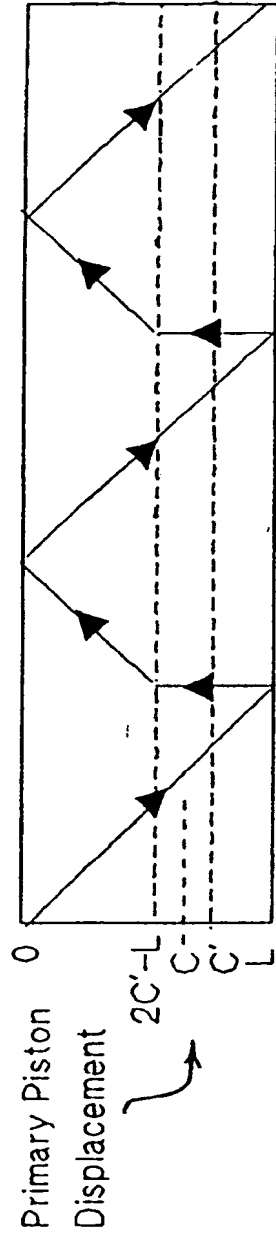


FIG 9A

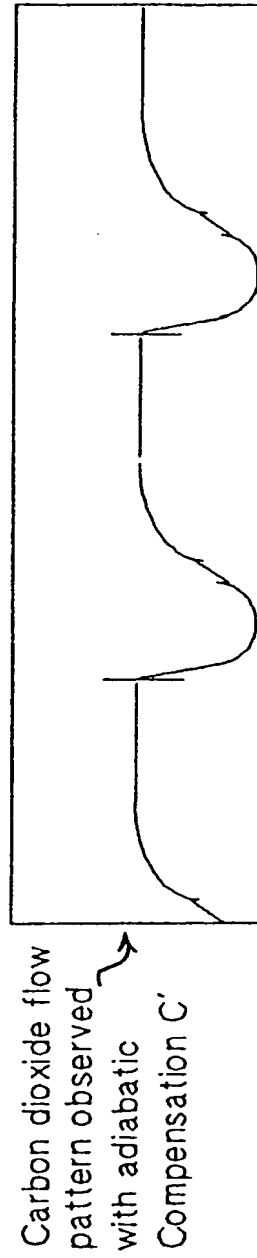


FIG 9B

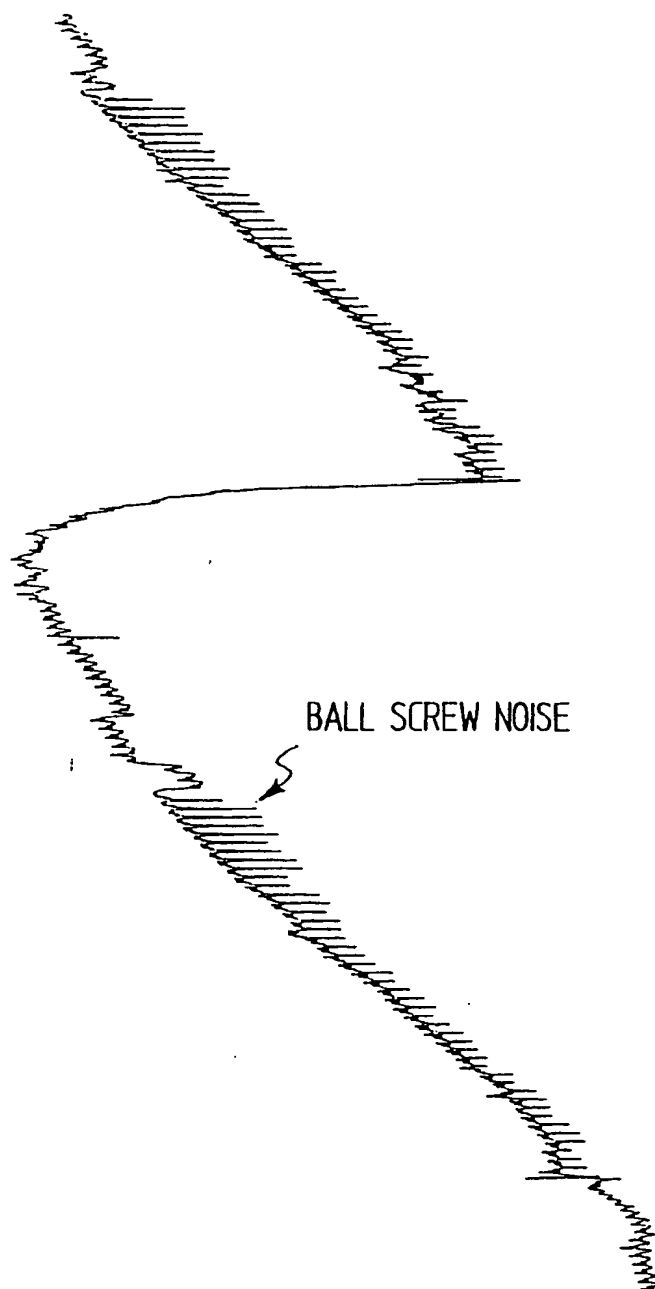


FIG 10

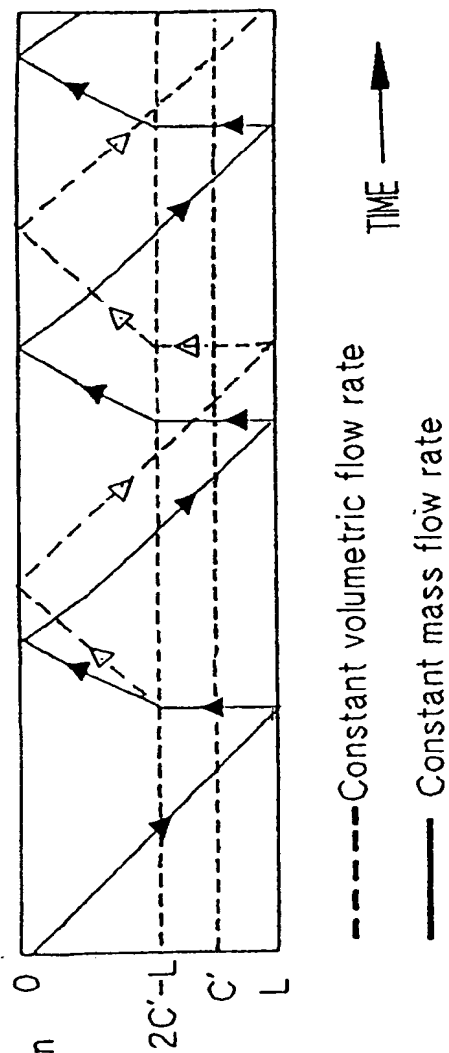


FIG 11A

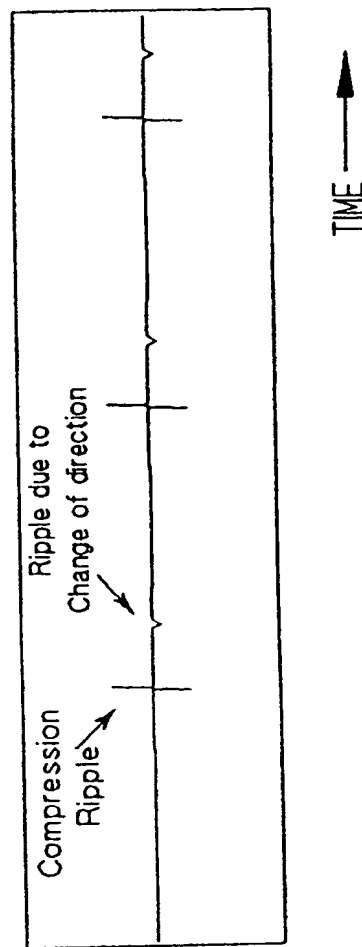


FIG 11B

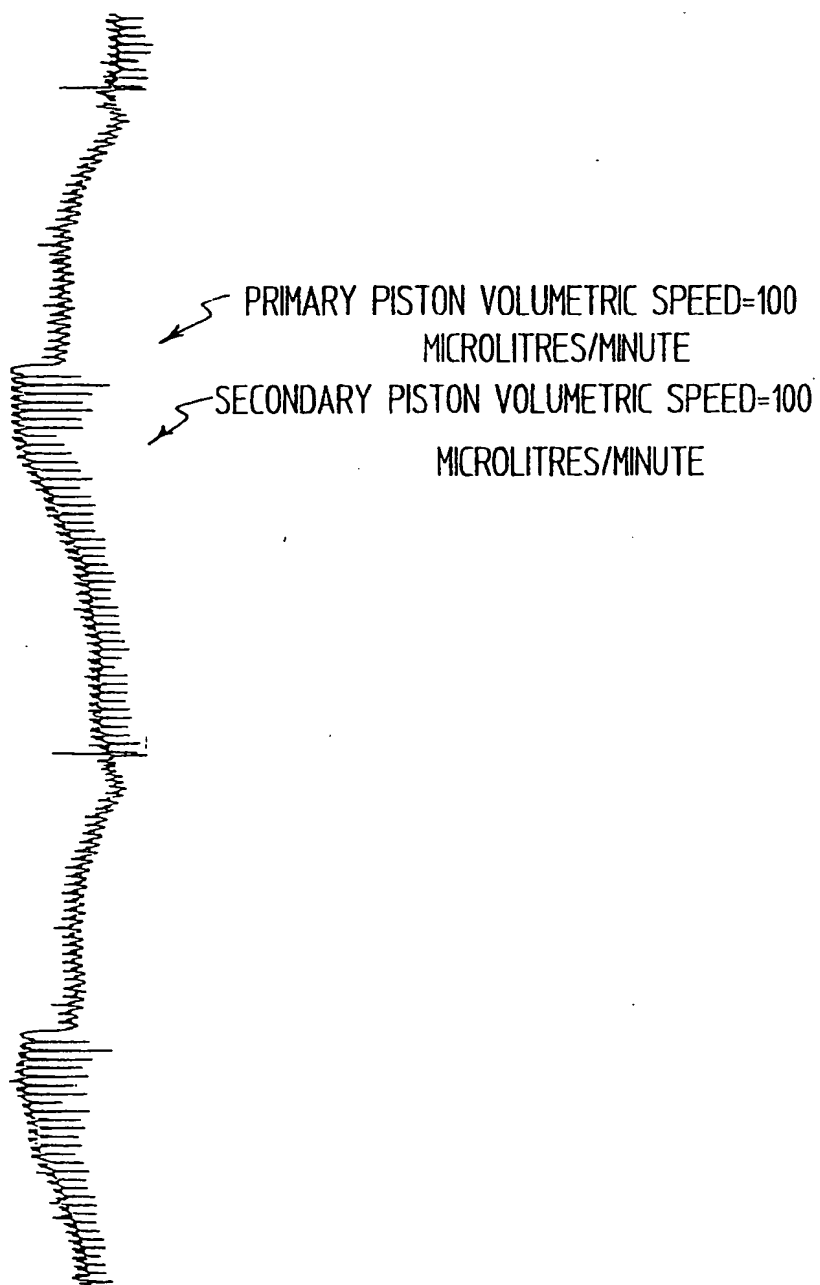


FIG 12

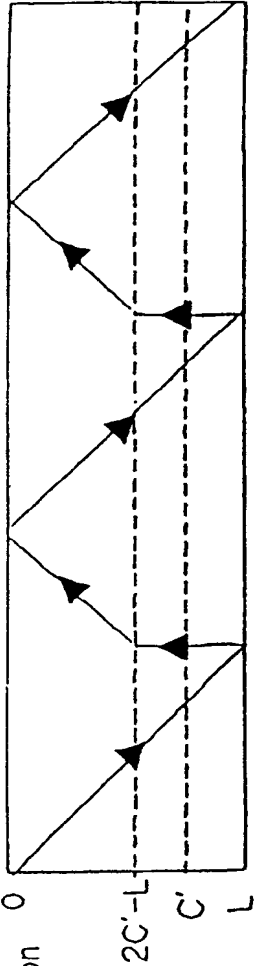


FIG 13A

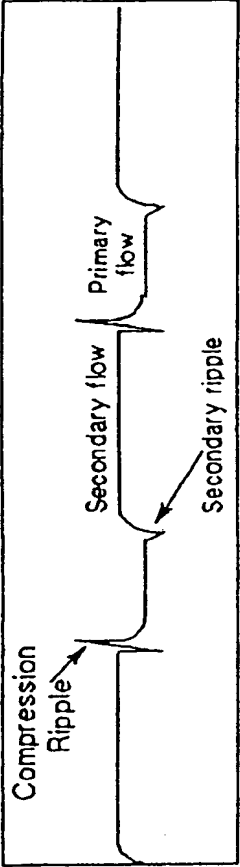


FIG 13B
Flow pattern for
leaky primary at
constant outlet pressure

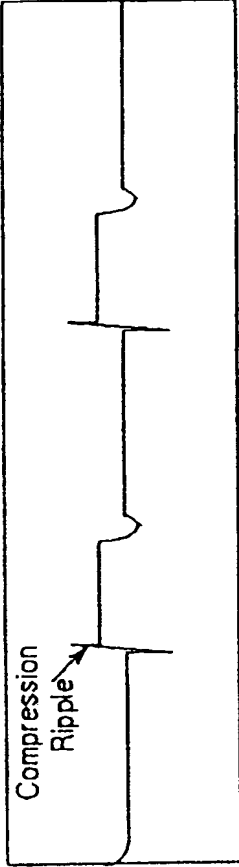


FIG 13C
Flow pattern for
leaky secondary at
constant outlet pressure

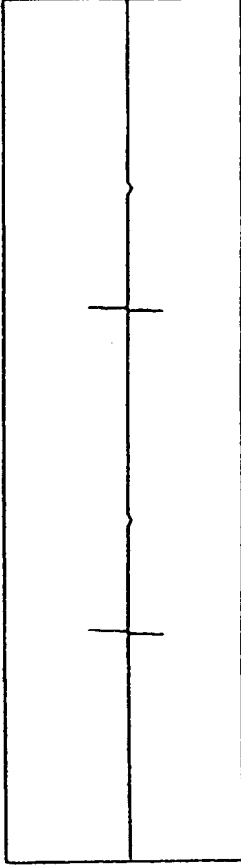


FIG 13D
PRESSURE & FLOW
PATTERN AFTER LEAK
CORRECTION

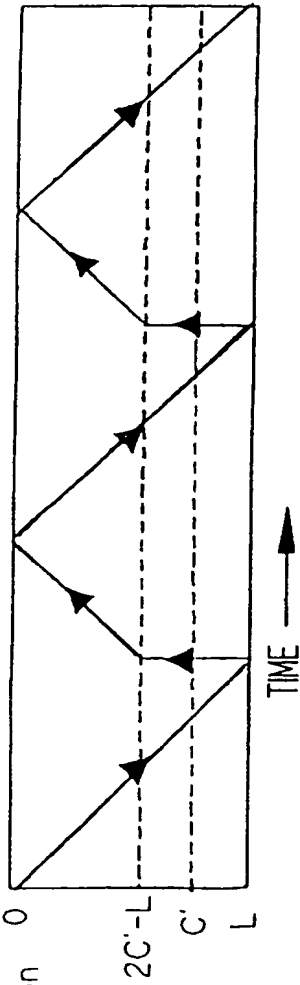
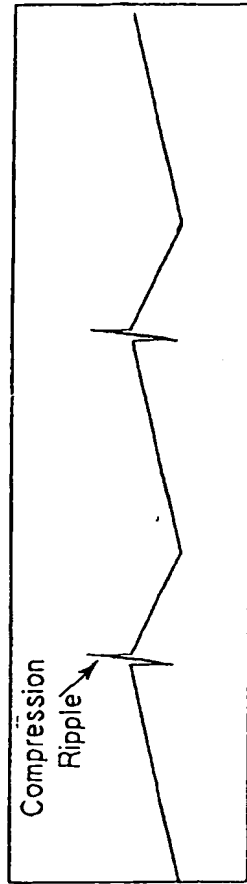
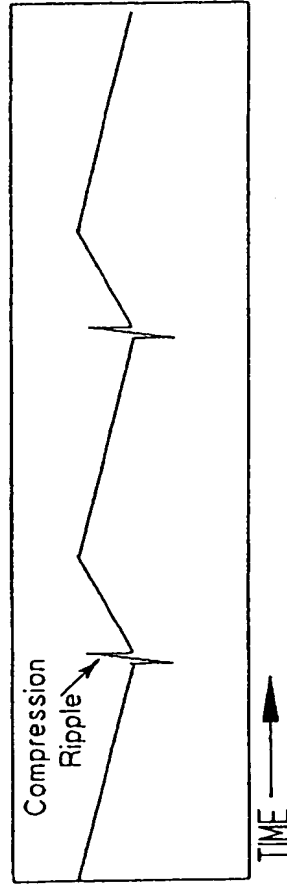


FIG 14A



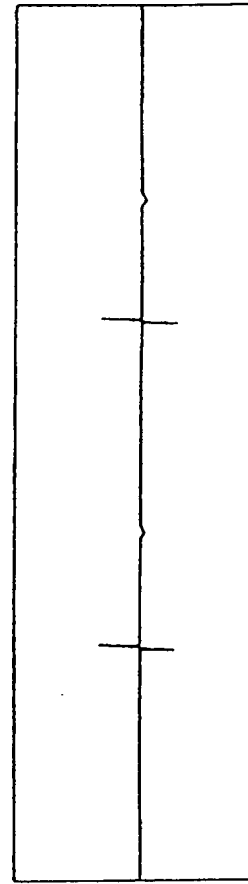
Pressure pattern for leaky primary using fixed restrictor

FIG 14B



Pressure pattern for leaky secondary using fixed restrictor

FIG 14C



Pressure and flow pattern after leak correction

FIG 14D

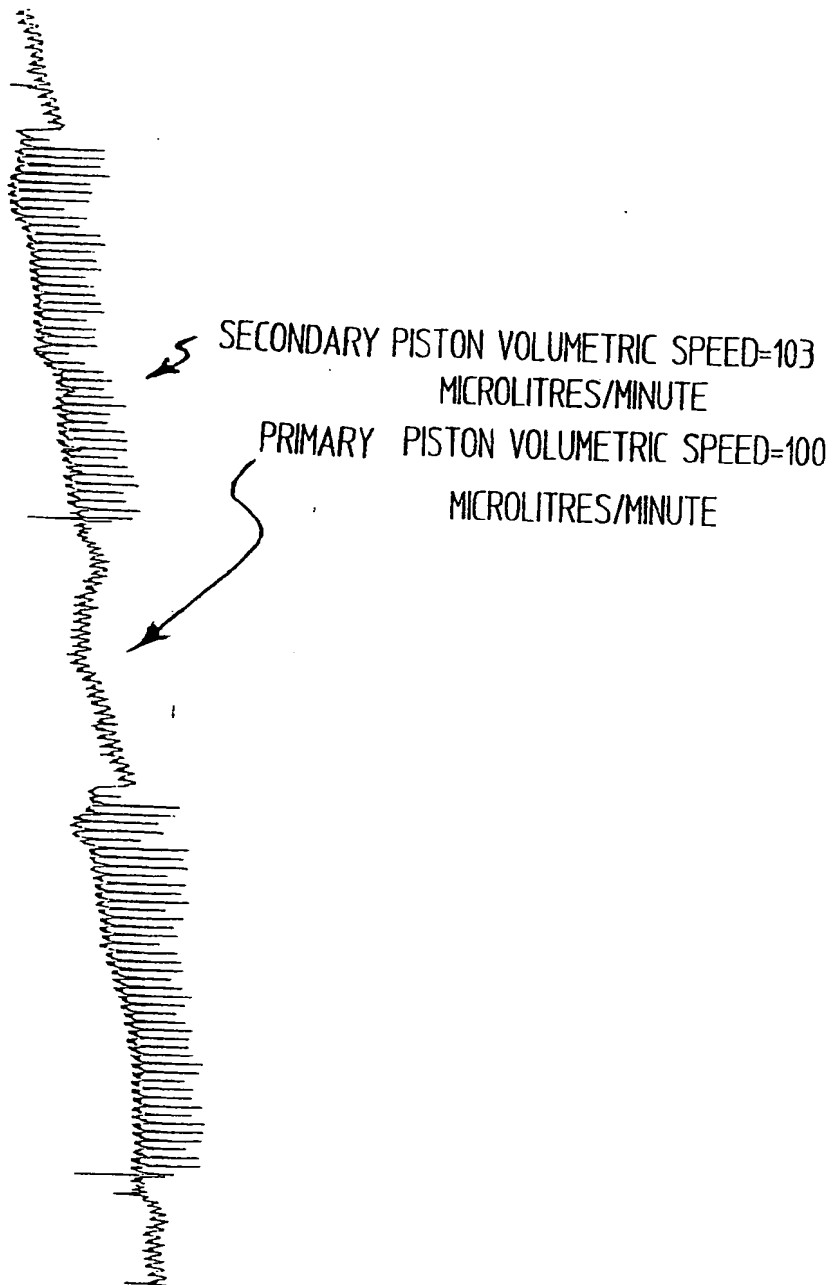


FIG 15

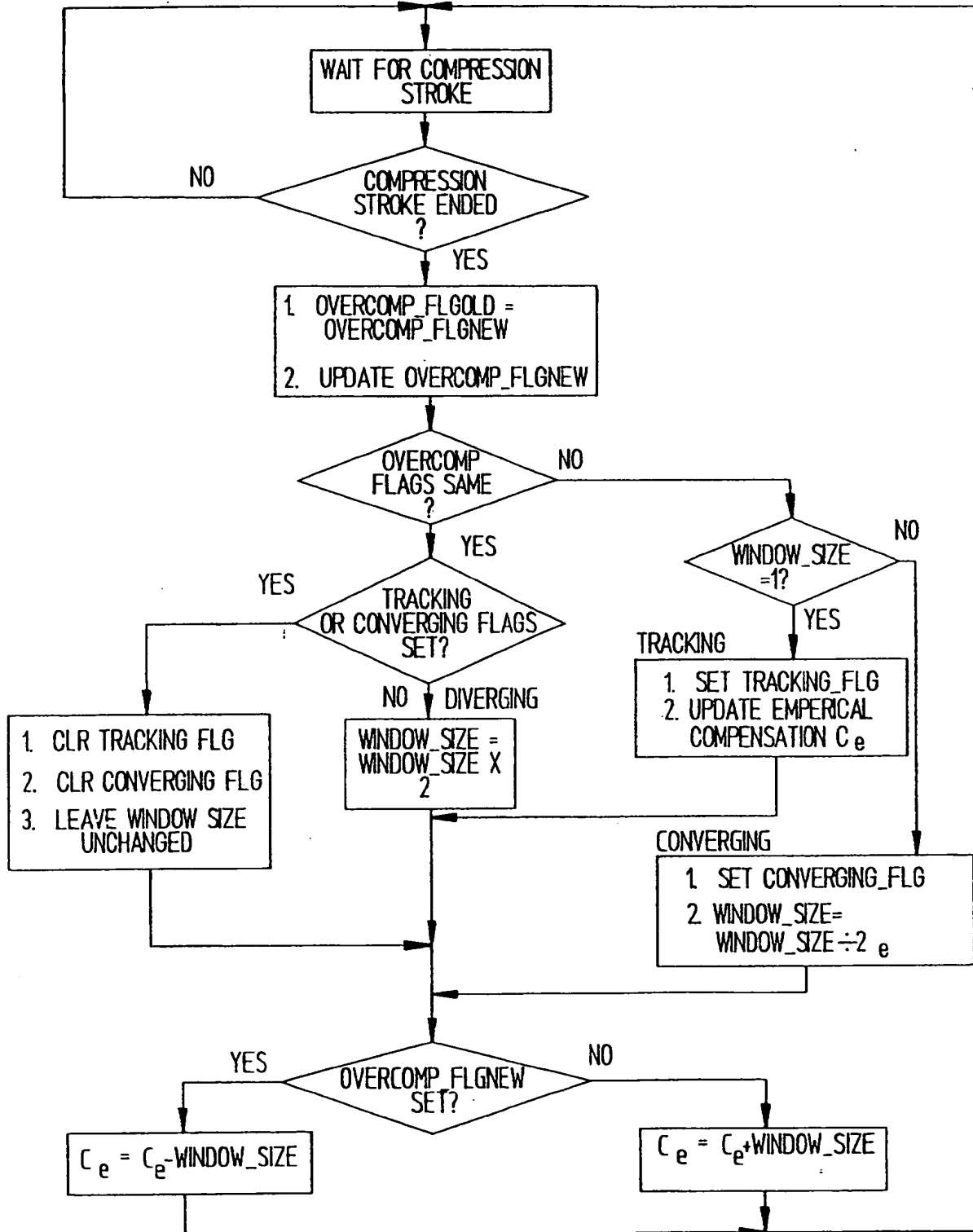


FIG 16



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EUROPEAN SEARCH REPORT

Application Number

EP 91 10 9005

DOCUMENTS CONSIDERED TO BE RELEVANT			
Category	Citation of document with indication, where appropriate, of relevant passages	Relevant to claim	CLASSIFICATION OF THE APPLICATION (Int. Cl.5)
X	FR-A-2 490 742 (BRUKER-ANALYTISCHE MESSTECHNIK) * page 1, line 1 - line 6 * * page 1, line 32 - page 2, line 12 * * page 3, line 21 - page 3, line 25 * * page 7, line 4 - page 13, line 35; figures *	4	F04B11/00
A	---	1,7,10	
A	EP-A-0 264 934 (HITACHI) * column 1, line 4 - line 9 * * column 3, line 26 - column 7, line 36; figures *	1,4,10	
A	---		
A	US-A-4 566 858 (TAKASHI AKIBA) * abstract * * column 1, line 29 - line 52 * * column 3, line 39 - column 4, line 53; figures *	1,3,4,7	
A	---		
A	FR-A-2 012 512 (FARBENFABRIKEN BAYER) * page 1, line 1 - line 8 * * page 1, line 33 - page 2, line 6 * * page 2, line 14 - page 3, line 24; figures *	2,5,6,10	TECHNICAL FIELDS SEARCHED (Int. Cl.5)
A	---		
A	EP-A-0 367 099 (BRUKER-FRANZEN ANALYTIK) * abstract * * page 1, line 21 - line 29 * * page 3, line 34 - line 39 * * page 3, line 53 - page 4, line 13; figure 1 *	1	F04B
The present search report has been drawn up for all claims			
Place of search THE HAGUE		Date of completion of the search 06 NOVEMBER 1991	Examiner ZIDI
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